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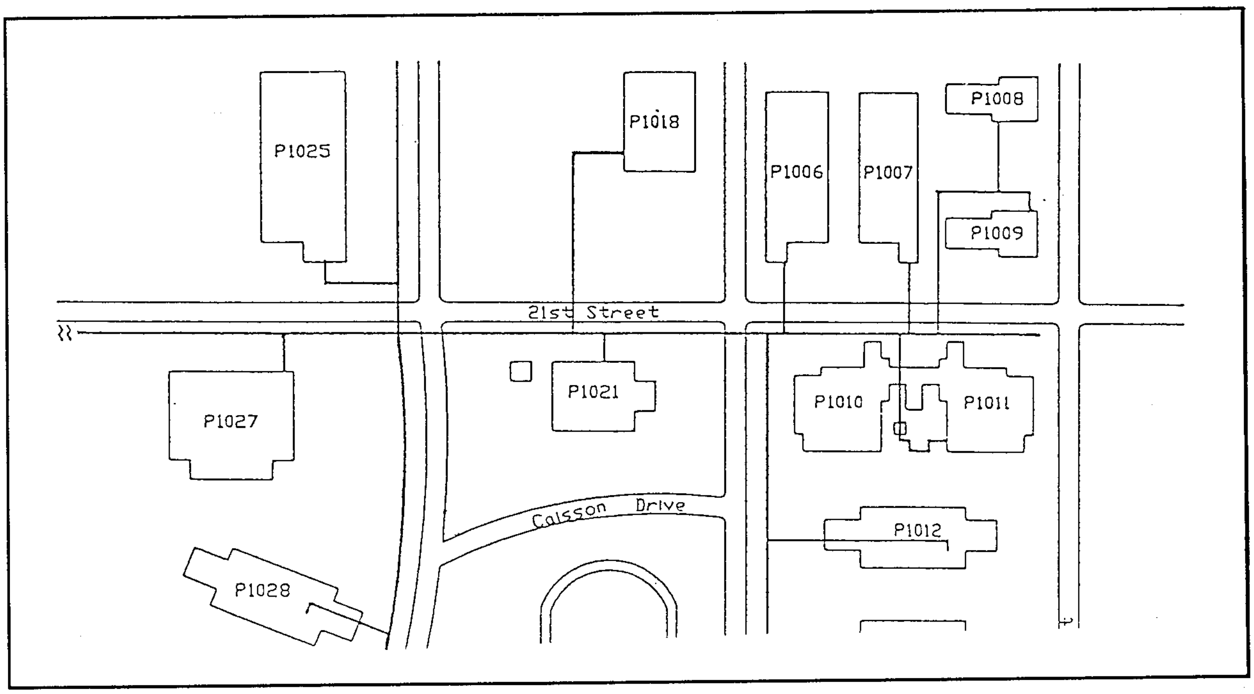
Alternative Refrigerant Performance

Field Test of a Nonchlorofluorocarbon Chiller at Fort Leonard Wood, MO

by
Chang W. Sohn, John J. Tomlinson, Nancy C. Herring, and Brian E. Boughton



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The U.S. Army Construction Engineering Research Laboratories (USACERL) and the U.S. Army Center for Public Works (USACPW) are doing research to find an efficient, alternative refrigerant for Army installations. The current project monitored the performance of a non-CFC (R-134a) centrifugal chiller at Fort Leonard Wood (FLW), MO. Performance of this chiller under field conditions was compared with the manufacturer's published ratings. Operational characteristics of the R-134a chiller were obtained by measuring electrical energy consumption, cooling delivered to the chiller cooling loop, and heat rejected by the condenser. Results indicated an average performance of approximately 0.68 kilowatts per ton (kW/ton) for the study period. The manufacturer's design projection was 0.73 kW/ton. The performance evaluation of the R-134a system shows that it is an efficient **addition to the FLW facility.**

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Foreword

This research was performed for the U.S. Army Center for Public Works (USACPW), under Project 4A162784AT45, "Energy and Energy Conservation"; Work Unit FE-XM4, "CFC Alternative Refrigerant Technologies." The USACPW technical monitors were Chris Irby, CECPW-EM, and Henry Gignilliat, Assistant Chief of Staff (Installation Management (ACS(IM)), DAIM-FDF-U/H.

The work was performed by the Energy and Utility Systems Division (FE), Infrastructure Laboratory (FL), U.S. Army Construction Engineering Research Laboratories (USACERL). The USACERL principal investigator was Dr. Chang W. Sohn. David Joncich is Division Chief, CECER-FE. Alan Moore is Acting Chief, CECER-FL. The USACERL technical editor was Agnes E. Dillon, Information Management Office.

Appreciation is expressed to the Directorate of Public Works staff at Fort Leonard Wood, MO, for their support to the project, the coordination provided by Roy McCarty was especially critical for the success of the project. The contribution by W. P. Levins of the Oak Ridge National Laboratory (ORNL) in the development of the instrumentation system also is appreciated.

LTC David J. Rehbein is Commander and Acting Director, USACERL. Dr. Michael J. O'Connor is Technical Director.

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1 Introduction

Background

The November 1992 Copenhagen Amendments to the Montreal Protocol call for an accelerated phaseout (January 1, 1996) of several ozone-depleting substances, including chlorofluorocarbon (CFC) refrigerants. In response to this international protocol, the U.S. Environmental Protection Agency (USEPA), as directed by the 1990 Clean Air Act Amendments (PL 101-549), proposed regulations (58 FR 15014) to implement the Copenhagen Amendments. Production of CFC refrigerants will stop permanently by the end of 1995. The 1993 Defense Authorization Act (PL 102-484, Sect. 326) prohibits Department of Defense (DOD) contracts awarded after June 1, 1993, from including a specification or standard that requires use of a Class I ozone-depleting substance or that can be met only through the use of such a substance (PL 102-484). CFC refrigerants are listed as Class I substances in the Clean Air Act Amendments of 1990 (PL 101-549, Sect. 602). The imminent need for alternative refrigerants and technologies for air-conditioning and refrigeration (AC/R) systems has spurred their development; however, the development of alternative refrigerants is still in its early stage (Sohn, Homan, and Herring, July 1994).

The U.S. Army has AC/R systems with a total cooling capacity of more than 1 million tons. Approximately 55 percent of the refrigerants used in these systems are CFC-based (Sohn, Homan, and Sliwinski 1992). A large number of chillers that currently use CFC refrigerants must be replaced or converted to operate with non-CFC refrigerants (or maintained until economically feasible to replace or convert).

To assist U.S. Army engineers in resolving these CFC issues, the U.S. Army Construction Engineering Research Laboratories (USACERL), with support from the U.S. Army Center for Public Works (USACPW), is working on a number of research programs to enable a successful phaseout of the CFC/HCFC (hydrochlorofluorocarbons) refrigerant-based AC/R systems from Army installations. In particular, USACERL and the Oak Ridge National Laboratory (ORNL) monitored the performance of a non-CFC centrifugal chiller installed at Fort Leonard Wood (FLW), MO, so its measured performance under field conditions could be compared with projected performance and provide guidelines for future implementation of R-134a systems.

Objectives

The objectives of this project were to measure the performance of a R-134a centrifugal chiller under field conditions and compare its field performance with the manufacturer's published ratings.

Approach

The R-134a chiller was instrumented with measuring devices so its performance, and not any system losses, was measured. This allowed measurement of the electrical energy consumption, cooling delivered to the chiller cooling loop, and heat rejected by the condenser for one cooling season. Monitoring of the temperature, flow rate, and electrical energy input provided the operational characteristics of the R-134a chiller. Operation and maintenance requirements were obtained from the contractor that serviced the R-134a chiller. A comparison then was made to the operation and maintenance requirements of the R-11 chiller.

Scope

The monitoring of this unit did not take into account system losses. The existing instrumentation on the unit chiller monitored the performance of the chiller alone. Because chiller manufacturers typically measure only the chiller performance instead of the entire system, modification of the chiller allowed for an adequate comparison of its performance to the manufacturer's projected performance.

Mode of Technology Transfer

The information in this report will be issued in a Public Works Technical Bulletin on the available options for a CFC phaseout program for field engineers. The results of this study also will be presented at the 1995 U.S. Army Corps of Engineers (USACE) Electrical and Mechanical Engineering Conference in a paper titled, "Field Performance of an R-134a Chiller at U.S. Army Facility."

2 System Configuration

Project History

In the spring of 1992 FLW replaced an R-11, 1,025 ton, low pressure centrifugal chiller with a 655.7 ton high pressure centrifugal chiller, R-134a. The R-134a chiller was installed next to the R-11 chiller in Building 1021. The R-11 chiller was approximately 22 years old and operated at 30 to 50 percent capacity most of the time. An R-134a chiller was selected to replace the R-11 chiller for several reasons.

- The age of the R-11 unit, the need for reduced capacity, and subsequent better efficiency indicated the need for a design change or replacement of the unit.
- The phaseout date of production and consumption of R-11 is rapidly approaching (January 1, 1996).
- No "drop-in" replacements for R-11 currently are available. R-134a could not have been used in the existing unit without major retrofit to the compressor and vessels because of the thermodynamic differences between R-11 and R-134a (see Figure 1).
- Technical Note (TN) 420-54-01 prohibits the procurement of ozone-depleting substances if suitable alternatives are available. R-11 is classified by the Clean Air Act Amendments (PL 101-549, Sect. 602) as a Class I ozone-depleting substance. R-134a is not classified as an ozone-depleting substance, and its production and consumption is not scheduled for phaseout.

The R-134a chiller ran for the cooling seasons in 1992 and in 1993, and its performance was monitored in the summer of 1993.

Load Description

The R-134a chiller services 18 buildings on the FLW installation. Buildings P1012, P1013, P1014, P1015, P1016, P1028, and P1029 are barracks with 40,640 square feet (sq ft) each (Figure 2). Buildings P1010, P1011, and P1027 are dining halls with 11,316 sq ft each. Buildings P1025, P1006, and P1007 are administration and supply

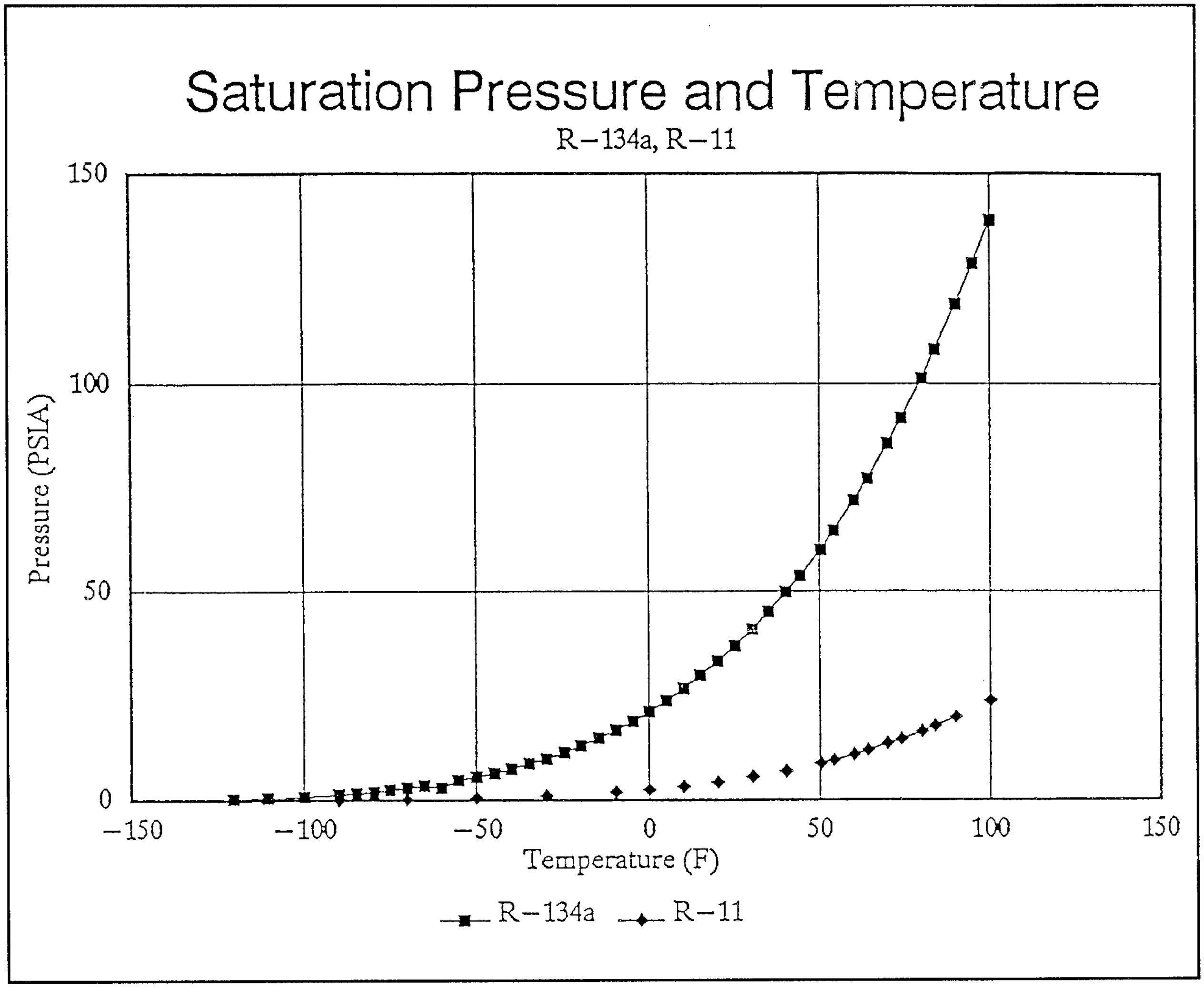

Figure 1. Saturation pressure and temperature of R-134a and R-11.

buildings, with 12,155 sq ft* each. The clinic, Building P1018, is 3,700 sq ft. Buildings P1008 and P1009 are each 6,163 sq ft. Building P1008 is an administration and classroom building, and Building P1009 is an administration and supply building. Two other buildings (5265 and 5400) not shown in Figure 2 also are cooled by the new chiller unit. Building 5265 is an administration and maintenance building with 4,346 sq ft; and Building 5400, also known as Brown Hall, has classrooms and is 112,480 sq ft. Therefore, the R-134a chiller provides cooling for a total of 487,745 sq ft.

Chiller Specifications

The chiller unit was designed according to USACE Construction Guide Specification 15650 (July 1992). The specifications particularly called for a centrifugal, liquid-

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* Metric conversion table is on page 27.

chilling package with refrigerant R-134a to provide 8,028,000 British thermal units (Btu) per hour of cooling while operating with the following parameters:

chilled water supply temperature	42 °F
chilled water return temperature	50 °F
condenser water leaving temperature	95 °F
condenser water entering temperature	85 °F
power input (maximum)	510 kW.

The chiller design chosen by the contractor to meet these specifications is summarized in Table 1.

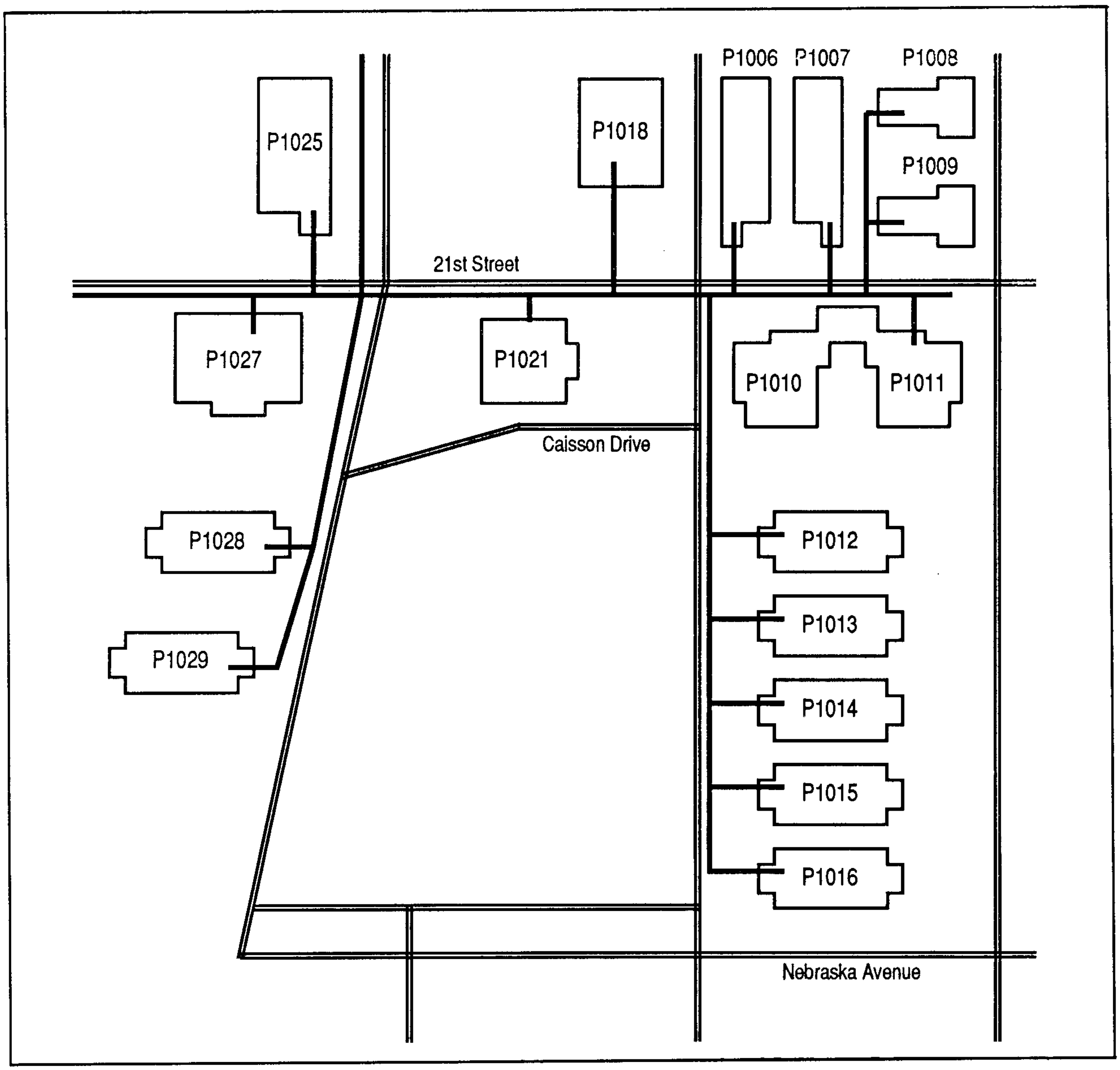

Figure 2. Building #1021 chiller load, Fort Leonard Wood.

Table 1. R-134a chiller design conditions.

Evaporator	Condenser	Motor
Number of passes = 2	Number of passes = 2	Electrical supply = 2400 VAC
EET* = 49.8 °F	ECT = 85.0 °F	LRA = 591 A
LET = °F	LCT = 93.0 °F	FLA = 127 A
ΔP = 11.26 psig	ΔP = 11.26 psig	Power consumption = 477 kW
Flow rate = 2007 gpm	Flow rate = 2400 gpm	
Suction temperature = 38.4 °F	Discharge temperature = 97.0 °F	

* EET = entering evaporator water temperature, ECT = entering condenser water temperature, LRA = locked rotor ampere, LET = leaving evaporator water temperature, LCT = leaving condenser water temperature, FLA = full load ampere, A = ampere, ΔP = pressure drop, VAC = volts alternating current, psig = pounds per square inch gauge, gpm = gallons per minute, kW = kilowatt.

3 Data Collection

Instrumentation Requirements

To determine the chiller performance (in kilowatts per ton) requires measurement of

- the electrical power to the chiller,
- the flow rate of chilled water through the evaporator,
- the entering evaporator water temperature (EET) and
- the leaving evaporator water temperature (LET).

Additional instrumentation to measure the tower water flow rate through the condenser, the entering condenser water temperature (ECT), and the leaving condenser water temperature (LCT) provides information needed to perform an energy balance on the chiller.

Chiller Instrumentation

Data collection combined the use of instrumentation provided as part of the original chiller as well as additional instrumentation ORNL installed to meet the objectives of this project.

One feature of the chiller was a McQuay MicroTech Unit Controller* (MUC) located on the chiller itself. The MUC is a microprocessor-controlled panel designed to monitor the chiller operating conditions, to regulate the compressor's capacity by operating the inlet vane positions, to sequence start-up and shutdown of the chiller, and to protect the compressor from operating conditions that could be damaging. With the MUC, water and refrigerant temperatures at both ends of the condenser and evaporator and refrigerant pressures can be monitored. The MUC also can monitor the electrical current to the chiller motor and the compressor oil pressure and temperature. The MUC allows the operator to monitor chiller parameters and to establish setpoints by using a keyboard located on the front panel and viewing the information presented on a light emitting diode (LED) readout. The MUC also can be controlled and monitored

* McQuay, Minneapolis, MN.

remotely through a personal computer (PC) by using software developed by the manufacturer.

Thermistors are used in the chiller to monitor compressor suction and discharge temperatures as well as temperature associated with the evaporator and condenser. The accuracy of the thermistor temperature measurements was ± 0.5 °F (Electric Power Research Institute 1983), which agrees with data provided by the manufacturer (SnyderGeneral/McQuay 1991). Pressure sensors used in the chiller are direct current output linear variable differential transducer (LVDT) type transducers with an accuracy of ± 0.3 percent at 300 pounds per square inch (psi). Electrical current through one leg of the motor circuit is monitored using a donut-type current transformer around one of the motor leads and a precision resistor. No published manufacturer's data on the accuracy of this current flow measuring technique was found.

Additional flowmeters were installed in the 8 in. chilled water and condenser water piping so flow rates could be measured. Based on good accuracy, lack of moving parts, and low pressure drop, full bore 8 in. vortex shedding flowmeters were chosen. These flowmeters work on the principle where vortices are produced alternately on either side of a bluff body as a fluid passes, and the frequency of generation of these vortices is proportional to the volume flow rate of the fluid. Pressure sensors downstream of the wedge-shaped bluff body inside the meter are used to determine the frequency of vortex generation. Based on the expected temperatures and viscosities of the condenser and chilled water, the meters were calibrated and used to measure flow rates to an accuracy of ± 0.8 percent of reading (Johnson Yokogawa 1992). The meters provided a pulse output with a frequency proportional to the volume flow rate. The volume of the water passing through the evaporator and condenser can be determined by counting the pulses for a period of time.

A watt/watt-hour transducer was installed across the leads of the chiller to determine the electrical power demand and electric energy consumption. Current transformers and potential transducers were used to match the nominal current draw of the chiller (about 200 amperes) and the voltage across the chiller leads (2400 volts [V]) with the input specifications of the watt/watt-hour transducer. This transducer has two outputs: a millivolt signal proportional to the instantaneous power draw of the chiller, and a contact closure that operates as the chiller consumes each unit of electrical energy. For this project, the contacts cycled once every time the chiller consumed 1.6 kilowatt hours (kWh) of electrical power. The contact closures were connected to a simple electrical circuit (resistor, voltage supply, contact switch) to provide a pulse each time the contacts closed.

Pulses from both flowmeters and the pulses and analog voltage signal from the watt/watt-hour transducer were recorded to tape at 3-minute intervals through a datalogger that had been programmed to read data channels and write to a datacassette tape.

Data Collection

The data collection procedure was designed to be robust so, if electrical power to the building were interrupted at any time, data would continue to be gathered at specified intervals. Before the initiation of data collection, the internal clocks of the PC and datalogger were synchronized and tested. The PC clock has a battery backup and the datalogger is fully powered by battery, so any power outage in Building 1021 would not affect the intervals and frequency of data collection. With the manufacturer's program in the PC, data on time of day, temperatures and pressures from each heat exchanger, oil pressures, system operating pressures, and chiller electrical current were collected every 10 minutes. With the datalogger, pulses that characterize condenser and evaporator water flow rates and electrical energy consumption were counted for 3-minute intervals, the accumulated pulses were written to the data tape, and the counters were zeroed to initiate the next counting interval. Additionally, the time of each scan was written to the data tape. The recorded data were sent weekly to ORNL. At ORNL, the raw data from the PC and the datalogger were combined and recorded in one Lotus 1-2-3 spreadsheet in 10-minute intervals for the 2-month cooling season. A paper copy and an electronic copy of the spreadsheet were sent to USACERL for review.

4 Chiller Performance

Data Reduction

For each 10-minute record, the collected data were used to calculate the average water flow-rates through the evaporator and condenser in gallons per minute, average chiller cooling load in tons, chiller heat rejection in tons, and chiller energy consumption in kilowatts. Other parameters such as kilowatt per ton and a heat balance on the chiller also were calculated.

Evaporator and condenser flows were computed from the pulse data using simple conversion factors (0.14455 gal per pulse minute and 0.14501 gal per pulse minute), respectively) to yield gallons per minute. The cooling load experienced by the chiller is computed using the evaporator water flow rate, specific heat, and temperature difference across the evaporator. Equation 1 gives the cooling in tons:

$$Q = (mC_p \Delta T) K_1 \qquad \text{[Eq 1]}$$

where Q = load delivered by the chiller, tons
m = chilled water flow rate, gpm
C_p = specific heat of water, 1 Btu/lb °F
ΔT = temperature difference between entering and leaving evaporator water temperature, °F
K_1 = conversion = (1 ft^3/7.48 gal) × (62.4 lb/ft^3) × (1 ton-min/200 Btu).

Similarly, the heat rejected by the condenser was calculated using the measured inlet and outlet water temperatures and flow rates through the condenser. Equation 1 was used, except with the following differences:

where Q = heat rejected, tons
m = condenser water flowrate, gpm
ΔT = temperature difference between entering and leaving condenser water temperature, °F
K_1 = conversion = (1 ft^3/7.48 gal) × (62.1 lb/ft^3) × (1 ton-min/200 Btu).

Electrical energy consumed by the chiller is converted from the millivolts measurement, provided by the transducer to kilowatts using the following expression:

$$\text{kW} = 9.6 \times (\text{number of pulses per 10 minutes}).$$

From the kilowatt data and the tons of cooling delivered data, the efficiency (kilowatts per ton) of the chiller is determined. Also, a heat balance is performed on the chiller that is simply the energy consumption plus the cooling load minus the heat rejected.

Performance of the R-134a Chiller

The performance of the R-134a chiller, in kilowatts per ton, depends on several parameters:

- leaving evaporator water temperature (LET)
- entering condenser water temperature (ECT)
- evaporator flow rate
- condenser flowrate
- the capacity of the chiller that is controlled by the position of the inlet vanes on the suction side of the compressor that opens and closes to permit the proper quantity of refrigerant to enter the wheel or impeller.

As the difference between the LET and ECT increases, the difference between the refrigerant suction and discharge pressures, or the pressure ratio, also increases. To prevent compressor surging in which refrigerant tends to flow backward through the compressor, the inlet vanes close to reduce the mass flow of refrigerant through the compressor. This reduces the capacity of the chiller. Thus, for a high speed centrifugal compressor in which the operation is controlled by movable inlet vanes, the relationship between the pressure lift, capacity, and operating temperature is complex. In part of this analysis, the range of some variables is limited, and the interdependence of no more than three other variables can be studied. Although this can be accomplished by examining performance data with fixed operating conditions, this method severely limits the number of data points for analysis. Consequently, the range of operating conditions is limited so a reasonable number of data records can be used for each analysis.

Using this method, the data from mid-July to about mid-August was examined to determine how the chiller performance (kilowatts per ton) was related to ECT and to capacity. The results of this analysis are shown in Figure 3. Only those records in which the LET is between 42.0 and 42.2 °F are shown. Because the LET setpoint was

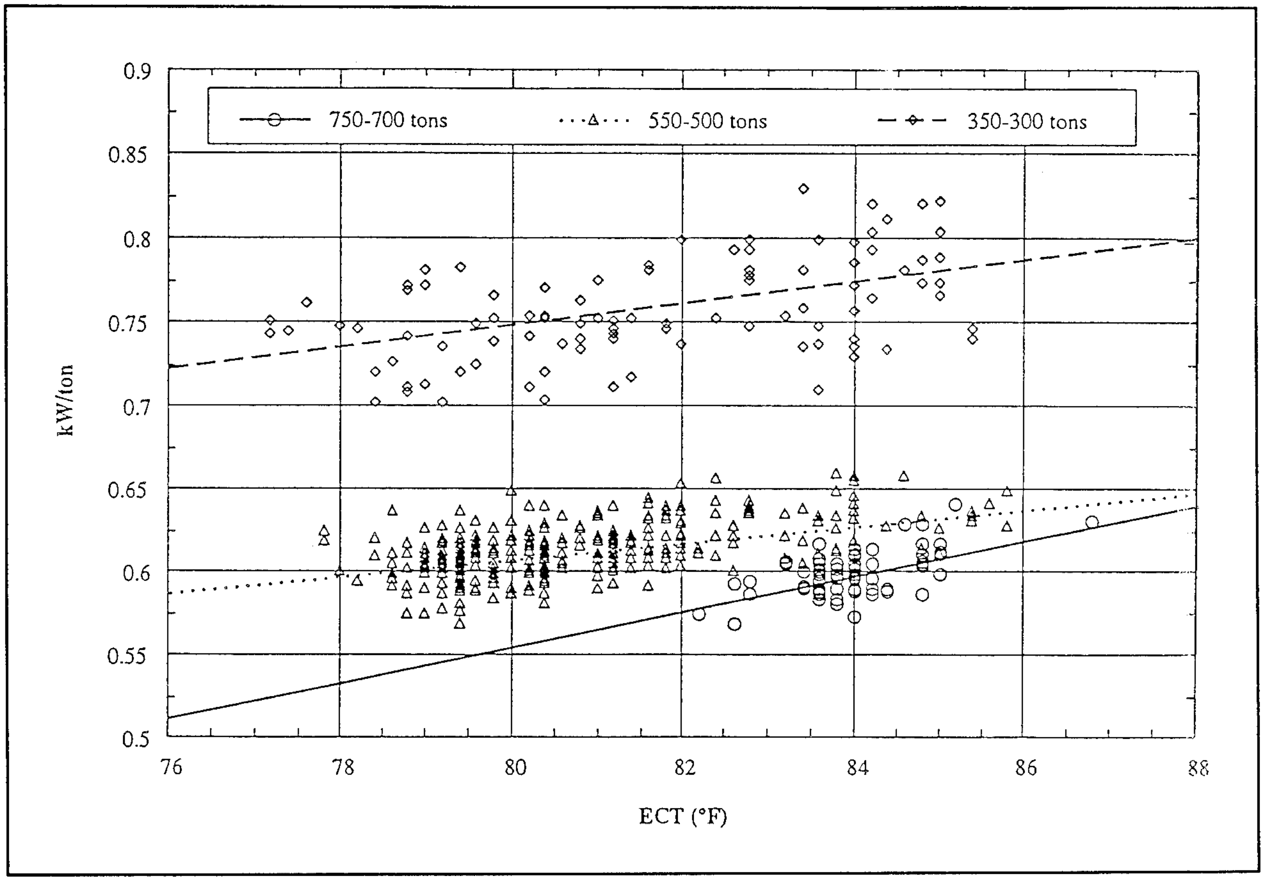

Figure 3. Chiller performance with LET between 42.0 and 42.2 °F for the first month of field study.

42 °F, the chiller usually operated at this temperature. The square data points shown represent records in which the chiller capacity is between 700 and 750 tons and the chiller is considered to be fully loaded. As the ECT increases, the kilowatts per ton tend to increase as shown by the regression curve. In a similar fashion, records with chiller capacities of 500 to 550 tons and 300 to 350 tons are grouped together. The performance trends are shown as the dashed and dotted lines. As the capacity of the chiller decreases, the kilowatts per ton increase and, even at low capacities, the chiller performance remains better than 0.9 kW per ton. Figure 4 shows this same analysis performed from mid-August through mid-September. The trends shown during mid-July to mid-August remain evident.

The entire data set was examined to determine how the average chiller performance varied with measured capacity. For this analysis, records were limited to those with the LET less than 43 °F, but no limit was set on the ECT. A cluster plot of these data is shown in Figure 5. The chiller performance improved as the capacity of the chiller increased. These data also show that the average chiller performance is best at the design capacity of approximately 660 tons. At lower capacities (about 30 percent of full load), the chiller performance is better than 1 kW per ton.

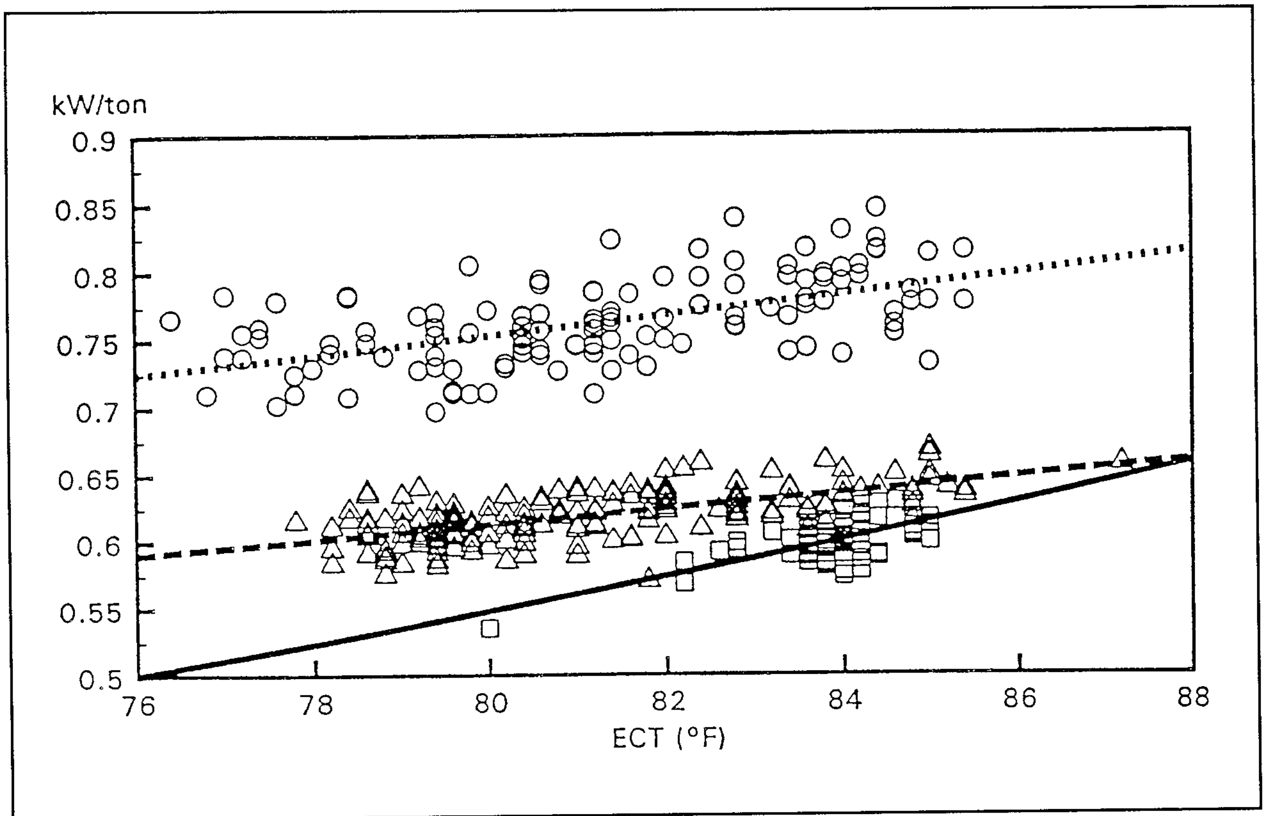

Figure 4. Chiller field performance with LET between 42 and 42.2 °F for the second month of study.

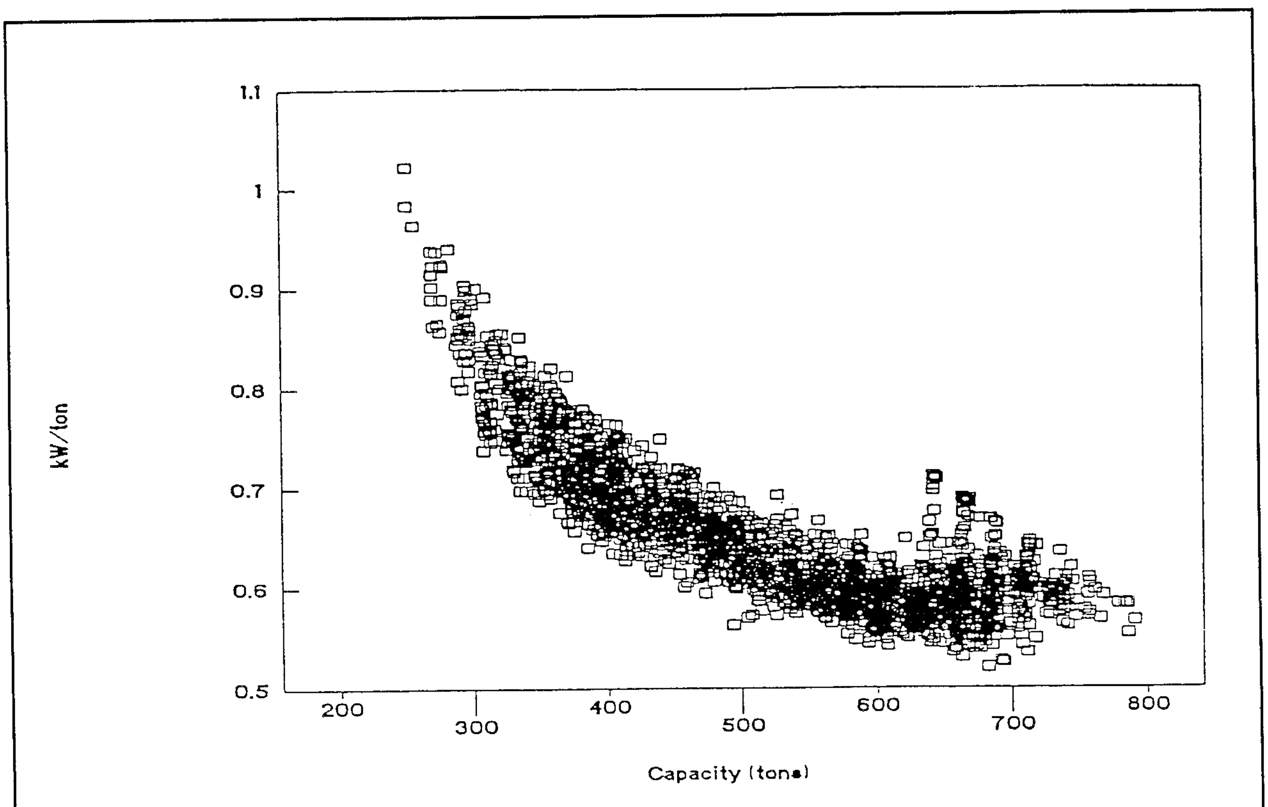

Figure 5. Chiller field performance with LET less than 43 °F and no limitations on ECT.

The effects of operating temperatures on chiller performance were examined for a typical hot summer day. The 24-hour period from 0800 Sunday, August 22 through 0800 Monday, August 23 was chosen. These data are shown in Figure 6. This particular 24-hour period was chosen for several reasons:

- The day was one of the hottest of the summer based on the outside dry bulb temperature reaching 96 °F on Sunday afternoon.
- The cooling demand is high during weekends, when the barracks are occupied.
- Examination of operating data for the chiller showed that the LWT and ECT as established by the tower and chiller controls remained uniform during this period so performance determination would not be affected by changing operating conditions.

Figure 6 shows that the cooling provided to the buildings increased from about 500 tons at 0800 on Sunday to approximately 700 tons at 2000. The maximum cooling load is experienced about 4 hours later than the peak daytime temperature. The location of the peak in the reject curve from the chiller indicates that the peak demands on the cooling tower occur 4 hours later than the hottest time of the day.

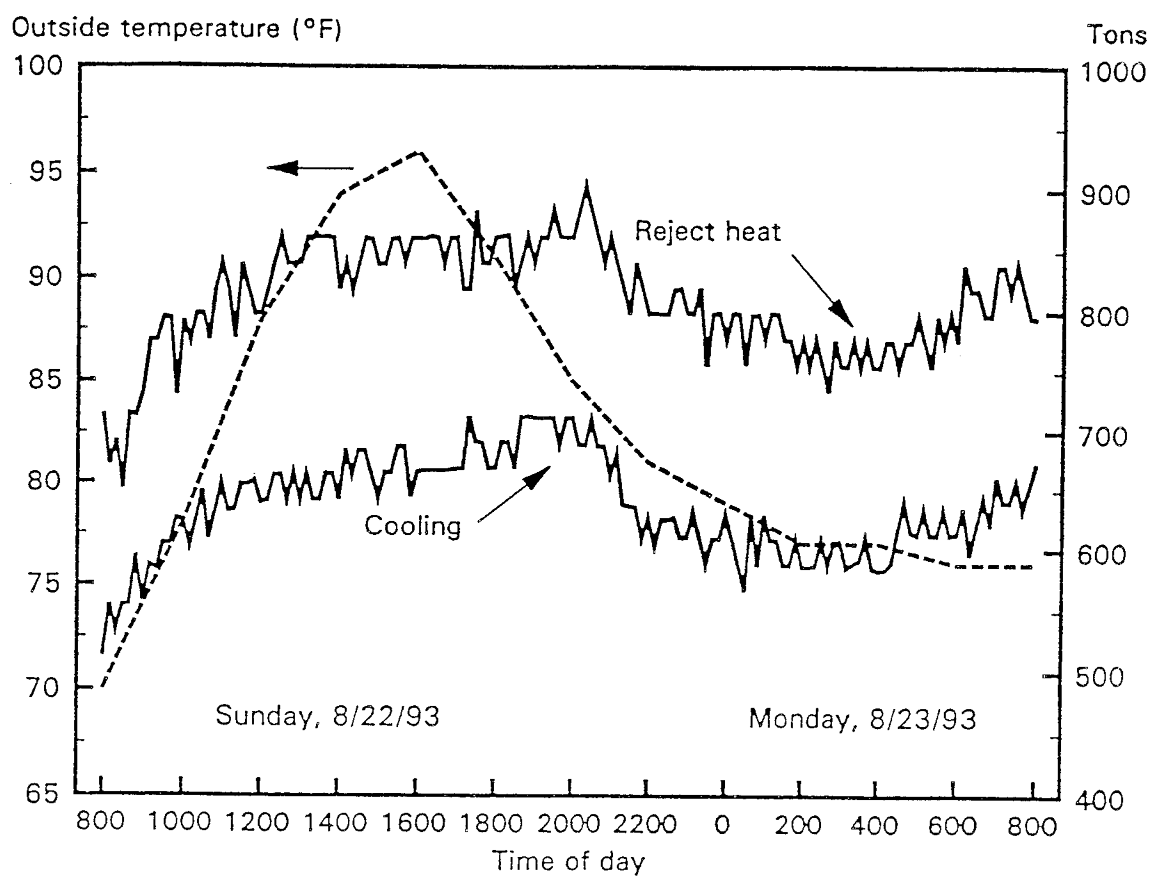

Figure 6. Chiller and system operating temperatures and performance for typical day.

Data from the same day was used to determine a typical value for fully loaded chiller performance as shown in Figure 7. The chiller LET remained constant over this period, and the EET changed in response to the cooling demand placed on the chiller. There is a direct relation between the calculated kilowatt per ton and the ECT so, even as the chiller load decreased early on Monday, a falling ECT improved the chiller efficiency. Figure 7 also shows that, at about 1400 on Sunday, the chiller was almost fully loaded and the 85 °F ECT ARI rating condition was attained. In this situation the chiller performance was about 0.7 kW per ton.

Discussion of Results

At the outset of the project, the sole manufacturer data on the performance of the chiller is that shown in Table 1. From this information and design operating conditions, the capacity of the chiller is calculated to be 653 tons, the heat rejection is 798 tons, and the performance is 0.73 kW per ton. A heat balance on the chiller shows that the thermal and electrical input to the chiller and the heat rejection from the condenser agree to within 1.2 percent.

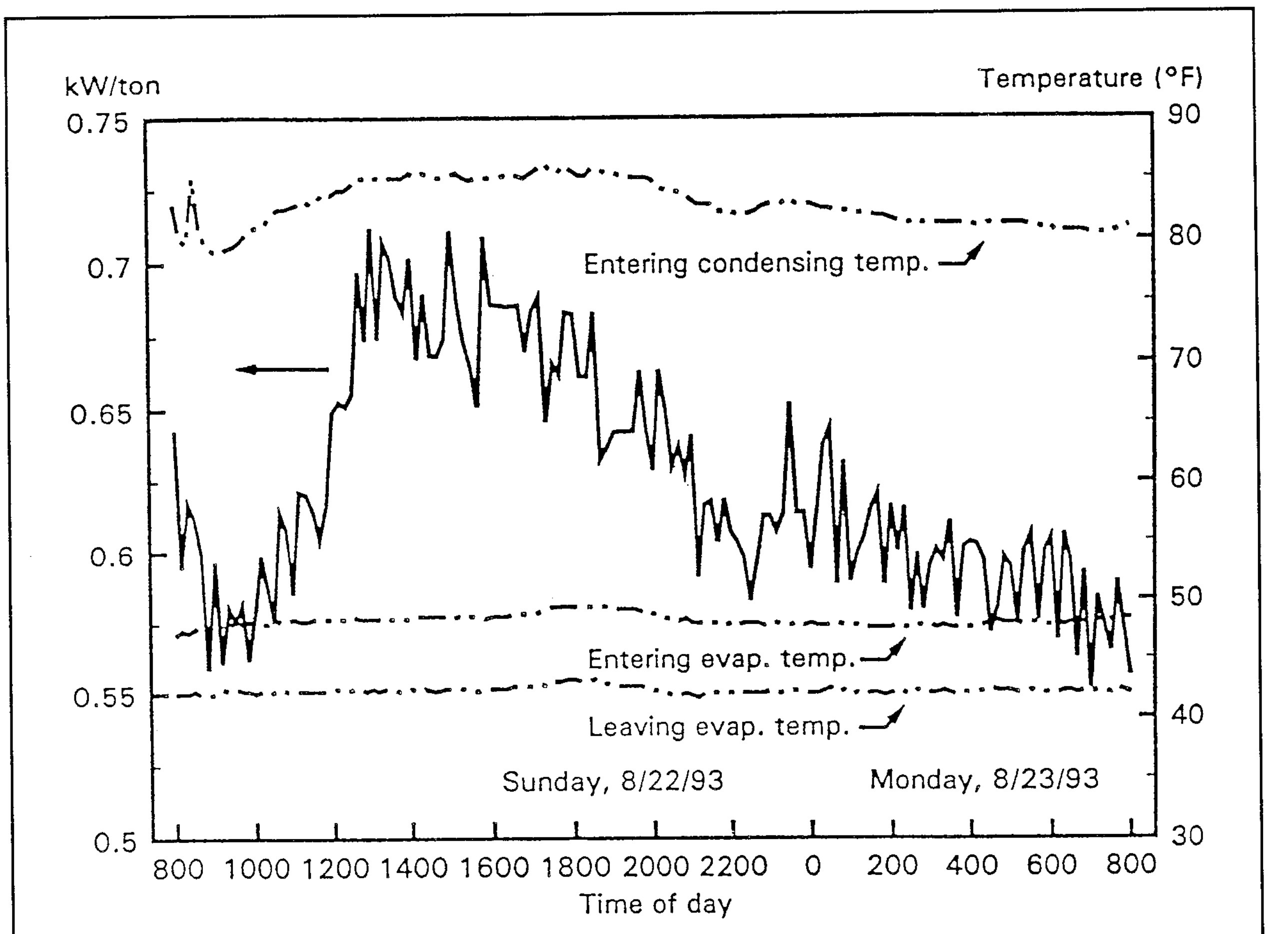

Figure 7. Chiller operating temperatures and performance for typical day.

Through field monitoring, the chiller generally is not being operated at the conditions shown in Table 1. The water flow rates through both the evaporator and the condenser are approximately 2,500 gpm; this is somewhat higher than the condenser design condition and significantly higher than the design evaporator condition. For the most part, field conditions differed from the design conditions, and comparisons of the measured performance with the expected performance required an estimation of manufacturer performance for actual field conditions. This information for each of the records in the experiment is not easily obtained because the performance of the chiller depends on a range of parameters. Furthermore, overall chiller specifications depend on the components used in the design of the chiller. For each chiller manufactured, a chiller barrel, condenser, impeller configuration, and gearing that best matches the design performance listed in the specifications is selected. Consequently, expected performance data on the chiller over the range of conditions found in the 2-month field study is not available in the manufacturer's published literature.

A simplified mathematical model of the chiller is constructed based on information from the manufacturer and estimates of the range of some of the operating parameters. In review of the field data, the following ranges were determined:

- condenser water flow rate varied from 2400 to 2800 gpm,
- ECT varied from 70 to 90 °F,
- evaporator flow rate varied from 2,000 to 2,600 gpm,
- LET varied from 42 to 44 °F,
- capacity varied from 300 to 700 tons.

The manufacturer provided performance data over the foregoing ranges of conditions. The conditions chosen for a parametric analysis are:

Evaporator	Condenser
flow rate = 2,000, 2,400, 2,600 gpm	flow rate = 2400, 2600, 2800 gpm
LET = 42, 44 °F	ECT = 70, 80, 90 °F
Capacity = 300, 500, 700 tons	

Through its design program, the manufacturer provided data on the performance of the chiller for most of the 162 unique combinations of these parameter values. Data were not available for evaporator flow rates exceeding 2,461 gpm and for condenser flow rates exceeding 2,500 gpm because the water velocities in each heat exchanger exceeded 10 feet per second (ft/s), and tube erosion is a concern at that flow rate. In addition, the manufacturer's program was unable to rate the chiller for condenser water temperatures higher than 85 °F. The manufacturer was unable to provide performance data for 74 combinations of the operating conditions, and this enabled

proceeding to development of a statistical performance model of the chiller. The first six columns in Table 2 show the operating conditions studied by the manufacturer, columns seven and eight are the manufacturer's estimate of power consumption and chiller performance.

A relatively simple, linear model—based on leaving evaporator temperature, evaporator flow rate, entering condenser temperature, condenser flow rate, capacity and electrical power—gives reasonably accurate estimates for chiller performance. The model is:

$$\begin{aligned} KWPERTON &= 5.47865\times10^{-1} - (1.215\times10^{-3})(LET) + (7.264\times10^{-6})(EVAPFLO) \\ &+ (3.612\times10^{-6}(CONDFLO) + (4.709\times10^{-3})(ECT) - (1.109\times10^{-3})(CAP) \\ &+ (1.140\times10^{-3})(PWR) \end{aligned}$$

where,

KWPERTON is the chiller performance (kW per ton)
LET is the leaving evaporator water temperature (°F)
EVAPFLOW is the chilled water flow rate (gpm)
CONDFLO is the condenser water flow rate (gpm)
ECT is the entering condenser water temperature (°F)
CAP is the chiller capacity (tons)
PWR is the chiller electrical power consumption (kW)

With the performance data provided by the manufacturer, this model produced an R^2 of 0.935 and an adjusted R^2 of 0.930, which indicates that about 93 percent of the variation in kilowatt per ton values could be accounted for by the 6-parameter model. Based on the fact that the sample size (74 observations) is much larger than the number of parameters, the adjusted R^2 value is regarded as a valid indicator of the "goodness" of the model. The model is used to calculate chiller performance as shown in column 9 of Table 2, and the relative difference between the model results and the manufacturer's estimates are shown in column 10. Except for those situations when the condenser temperature is 70 °F and the chiller is fully loaded, the model and the manufacturer's results agree to within approximately 5 percent. Although additional data from the manufacturer combined with development of a more sophisticated model might improve agreement between the model and the manufacturer's data, little would be gained in terms of improving the comparison of model results and field experimental data. Instrumentation inaccuracies combined with uncertainties generated by the performance calculations produce results that are valid to within ±11 percent as shown in the Appendix. Therefore, using the existing model to compare field results with manufacturer's projections is reasonable.

Table 2. Comparison of manufacturer's performance estimate with mathematical model.

Point No.	LET* (°F)	Evap (gpm)	Cond (gpm)	ECT (°F)	Capacity (tons)	Power (kW)	Manufacturer (kW/ton)	Model (kW/ton)	Error (%)
1	42	2007	2400	70	700	399.7	0.571	0.530	7.3
2	42	2007	2400	80	685.2	460.5	0.672	0.663	1.5
3	42	2007	2500	70	700	397.6	0.568	0.528	7.2
4	42	2007	2500	80	685.9	459.6	0.670	0.661	1.4
5	42	2007	2500	85	656.9	476.3	0.725	0.735	1.4
6	42	2400	2400	70	700	400.4	0.572	0.534	6.8
7	42	2400	2400	80	682.8	460.2	0.674	0.667	1.0
8	42	2400	2500	70	700	398.3	0.569	0.532	6.7
9	42	2400	2500	80	683.5	458.6	0.671	0.665	0.9
10	42	2400	2500	85	654.2	474.9	0.726	0.740	1.9
11	44	2007	2400	70	700	389.9	0.557	0.516	7.4
12	44	2007	2400	80	700	462.7	0.661	0.646	2.3
13	44	2007	2500	70	700	387.8	0.554	0.514	7.3
14	44	2007	2500	80	700	460.6	0.658	0.644	2.2
15	44	2007	2500	85	686.8	487.6	0.710	0.713	0.4
16	44	2400	2400	70	700	390.6	0.558	0.520	6.9
17	44	2400	2400	80	700	463.4	0.662	0.649	1.9
18	44	2400	2500	70	700	389.2	0.556	0.518	6.8
19	44	2400	2500	80	700	461.3	0.659	0.648	1.8
20	44	2400	2500	85	684.1	486.4	0.711	0.718	0.9
21	42	2461	2500	70	700	398.8	0.570	0.532	6.6
22	42	2461	2500	80	683.2	458.6	0.671	0.666	0.8
23	42	2461	2500	85	653.9	474.9	0.726	0.741	2.0
24	44	2461	2500	70	700	389.1	0.556	0.519	6.7
25	44	2461	2500	80	700	461.8	0.660	0.649	1.7
26	44	2461	2500	85	683.7	486.4	0.711	0.718	0.9
27	42	2007	2400	70	499.5	303.7	0.608	0.642	5.6
28	42	2007	2400	80	499.5	357.6	0.716	0.751	4.8
29	42	2007	2400	85	499.5	385.1	0.771	0.806	4.5
30	44	2007	2400	70	499.5	298.2	0.597	0.634	6.1
31	44	2007	2400	80	499.5	353.1	0.707	0.743	5.1
32	44	2007	2400	85	499.5	380.6	0.762	0.798	4.8
33	42	2007	2500	70	499.5	302.7	0.606	0.646	6.0
34	42	2007	2500	80	499.5	356.6	0.714	0.754	5.2
35	42	2007	2500	85	499.5	384.1	0.769	0.809	4.8
36	44	2007	2500	70	499.5	297.2	0.595	0.637	6.5
37	44	2007	2500	80	499.5	352.1	0.705	0.746	5.5
38	44	2007	2500	85	499.5	379.1	0.759	0.801	5.1
39	42	2400	2400	70	499.5	304.2	0.609	0.641	5.8
40	42	2400	2400	80	499.5	358.1	0.717	0.750	5.0
41	42	2400	2400	85	499.5	385.1	0.771	0.805	4.7
42	44	2400	2400	70	499.5	298.7	0.598	0.633	6.3
43	44	2400	2400	80	499.5	353.1	0.707	0.742	5.3
44	44	2400	2400	85	499.5	380.6	0.762	0.797	4.9
45	42	2400	2500	70	499.5	303.2	0.607	0.645	6.3
46	42	2400	2500	80	499.5	357.1	0.715	0.753	5.4
47	42	2400	2500	85	499.5	384.1	0.769	0.808	5.0
48	44	2400	2500	70	499.5	297.7	0.596	0.636	6.7

Table 2. Continued.

Point No.	LET* (°F)	Evap (gpm)	Cond (gpm)	ECT (°F)	Capacity (tons)	Power (kW)	Manufacturer (kW/ton)	Model (kW/ton)	Error (%)
49	44	2400	2500	80	499.5	352.1	0.705	0.746	5.7
50	44	2400	2500	85	499.5	379.6	0.760	0.800	5.3
51	42	2007	2400	70	302.4	233.8	0.773	0.781	1.1
52	42	2007	2400	80	302.4	272.2	0.900	0.872	3.1
53	42	2007	2400	85	302.4	291.5	0.964	0.917	4.8
54	44	2007	2400	70	302.4	231.0	0.764	0.775	1.5
55	44	2007	2400	80	302.4	270.0	0.893	0.867	2.9
56	44	2007	2400	85	302.4	289.7	0.958	0.913	4.7
57	42	2007	2500	70	302.4	233.2	0.771	0.784	1.4
58	42	2007	2500	80	302.4	271.9	0.899	0.875	2.8
59	42	2007	2500	85	302.4	291.2	0.963	0.920	4.6
60	44	2007	2500	70	302.4	230.4	0.762	0.778	1.9
61	44	2007	2500	80	302.4	269.7	0.892	0.870	2.6
62	44	2007	2500	85	302.4	289.4	0.957	0.916	4.4
63	42	2400	2400	70	302.4	233.8	0.773	0.781	1.3
64	42	2400	2400	80	302.4	272.2	0.900	0.872	3.0
65	42	2400	2400	85	302.4	291.5	0.964	0.917	4.7
66	44	2400	2400	70	302.4	231.0	0.764	0.775	1.7
67	44	2400	2400	80	302.4	270.3	0.894	0.867	2.8
68	44	2400	2400	85	302.4	290.0	0.959	0.913	4.6
69	42	2400	2500	70	302.4	233.2	0.771	0.784	1.6
70	42	2400	2500	80	302.4	271.9	0.899	0.875	2.7
71	42	2400	2500	85	302.4	291.2	0.963	0.920	4.4
72	44	2400	2500	70	302.4	230.4	0.762	0.778	2.1
73	44	2400	2500	80	302.4	269.7	0.892	0.870	2.5
74	44	2400	2500	85	302.4	289.4	0.957	0.916	4.3

*LET = leaving evaporator water temperature, Evap = evaporator flow rate, Cond = condenser flow rate, Manufacturer = manufacturer's results, Model = mathematical model results.

In Figure 8, calculated field performance determined from measured data is compared with chiller performance projections from the statistical model. If the two agreed completely, the scatter would lie along the line shown. Error bars are added to this line based on the instrumentation accuracies. As can be seen, the width of the error bars depends on the magnitude of the kilowatt per ton measurement. Virtually all of the kilowatt per ton calculated from field measured data fall within the kilowatt per ton calculated from the model. Higher measurement accuracy, particularly with temperatures for small temperature changes as found across evaporators and condensers, is important to improve the significance of this comparison.

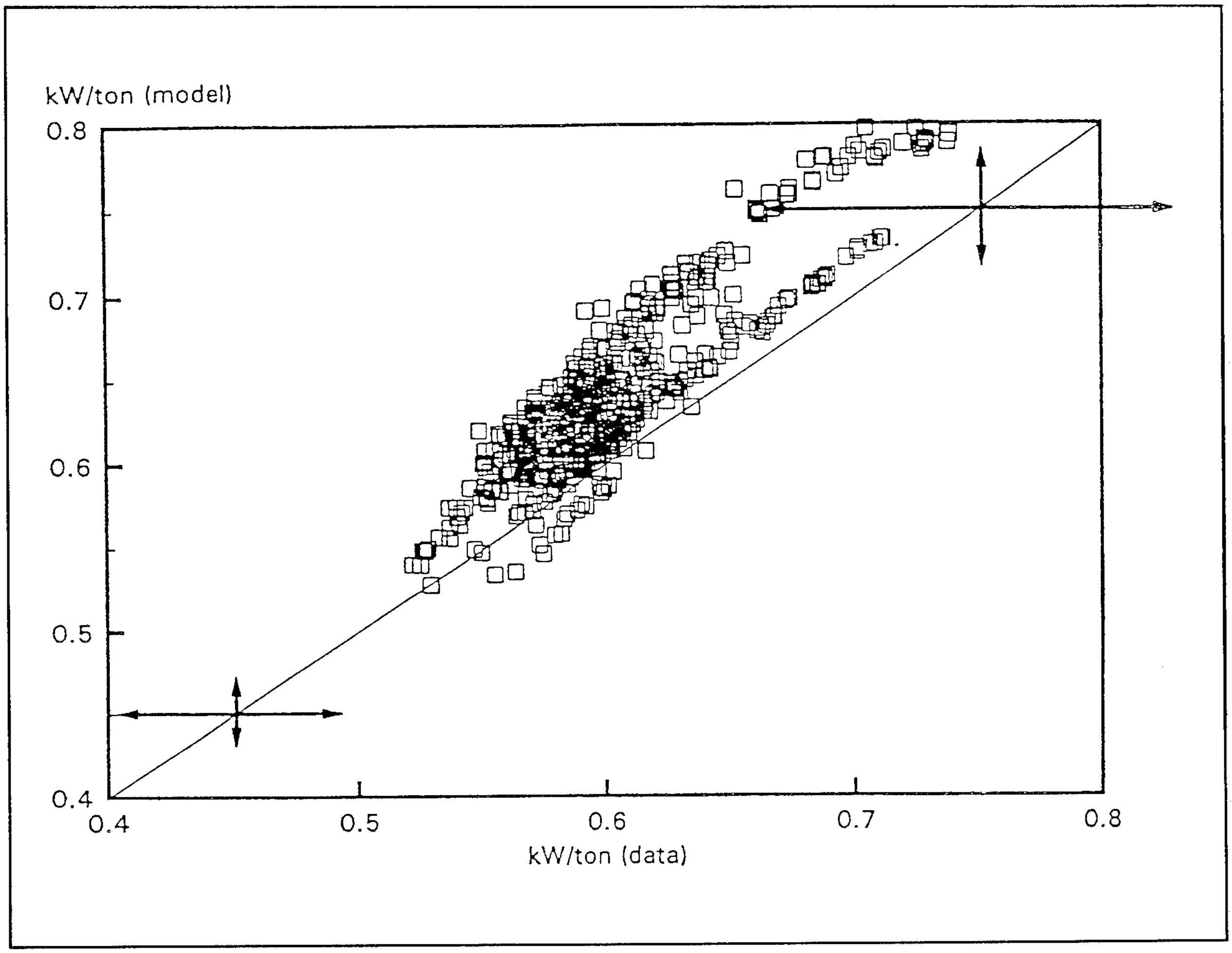

Figure 8. Comparison of chiller model with experimental data.

5 Conclusions

A 2-month field study determined that the R-134a chiller was efficient. Results of this study showed that the average performance of the chiller over the study period was 0.68 kW per ton (coefficient of performance = 5.2) ±11 percent. The uncertainty in the measured chiller performance is a result of instrumentation inaccuracies and uncertainties in the performance calculations.

Expected performance data on the chiller were obtained from the manufacturer. Multivariate regression was applied to these data, and a statistical model of the chiller performance good to within approximately 5 percent was developed. Using this model, the average predicted chiller performance for the 2-month monitoring period was calculated to be 0.75 kW per ton ±5 percent. The experimental and projected results agree with the accuracy of the instruments and model, and the results show that the R-134a chiller is an energy efficient and environmentally sound addition to the FLW facility.

Metric Conversion Table

1 in.	=	25.4 mm
1 ft	=	0.305 m
1 sq ft	=	0.093 m^2
1 lb	=	0.453 kg
1 gal	=	3.78 L
1 psi	=	6.894 Pa
°F	=	(°C x 1.8) +32
Btu/h	=	0.293 W
1 kWh	=	56.868 Btu
1 ft^3	=	0.028 m^3
tons	=	3516.8 W
psig	=	6894.7 Pa
1 gpm	=	0.063 L/s

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Appendix: Uncertainty Analysis

Uncertainty in Capacity Determination

The capacity of the chiller is a function of several variables:

$$Q = Q(m, C_p, T_1, T_2) \qquad \text{[Eq A1]}$$

where m = mass flow rate
C_p = specific heat of water
T_1 = entering chiller temperature
T_2 = leaving chiller temperature
Q = capacity

Assume that the flow rate, specific heat, and temperatures are constant over a sampling period, then:

$$Q = mC_p\,(T_1 - T_2) \qquad \text{[Eq A2]}$$

The uncertainty in Q can be calculated from the following relation (Kline and McClintock 1953).

$$dQ = \sqrt{\left(\frac{\partial Q}{\partial m}dm\right)^2 + \left(\frac{\partial Q}{\partial C_p}dC_p\right)^2 + \left(\frac{\partial Q}{\partial T_1}dT_1\right)^2 + \left(\frac{\partial Q}{\partial T_2}dT_2\right)^2} \qquad \text{[Eq A3]}$$

Performing the operations in [Eq A3] and dividing by [Eq A2], the relative uncertainty in Q is:

$$\frac{dQ}{Q} = \sqrt{\left(\frac{dm}{m}\right)^2 + \left(\frac{dC_p}{C_p}\right)^2 + 2\left(\frac{dT}{T_1 - T_2}\right)^2} \qquad \text{[Eq A4]}$$

The accuracy of the vortex-shedding flowmeters in the chiller water line is ±0.8 percent of the reading for the pulse output that is used on this project (Johnson Yokogawa 1992). The uncertainty in the specific heat of water is assumed to be zero. The uncertainty in temperature measurements from the chiller is taken to be ±0.5 percent (Electric Power Research Institute 1983; SnyderGeneral/McQuay 1991). Based on these uncertainties and a typical 6 °F temperature change across the evaporator, the uncertainty in capacity from [Eq A4] is ±11 percent.

Uncertainty in Performance Determination

The uncertainty in chiller performance is a combination of the uncertainty in capacity, Q (see equation A4), and the uncertainty in the average electrical power consumption. Following an analysis similar to that for capacity, the relative uncertainty in chiller performance is:

$$\frac{dK}{K} = \sqrt{\left(\frac{dQ}{Q}\right)^2 + \left(\frac{dP}{P}\right)^2} \qquad \text{[Eq A5]}$$

where K = performance, kW per ton
Q = capacity, tons
P = electrical power, kW

The uncertainty in chiller capacity is ±11 percent. The uncertainty in the watt/watt-hour transducers used for the project is ±0.15 percent. Substituting these two uncertainties into [Eq A5] yields a performance uncertainty of ±11 percent. This result is anticipated because the uncertainty in the power measurement is much smaller than the uncertainty in the capacity measurement.

Abbreviations and Acronyms

A	ampere
AC/R	air-conditioning and refrigeration
ARI	Air-Conditioning and Refrigeration Institute
Btu	British thermal unit
CFC	chlorofluorocarbon
ΔP	pressure drop
DPW	Directorate of Public Works
ECT	entering condenser water temperature
EET	entering evaporator water temperature
FLW	Fort Leonard Wood
gal	gallon
gpm	gallons per minute
HCFC	hydrochlorofluorocarbon
in.	inch
kW	kilowatt
kWh	kilowatt hours
lb	pound
LCT	leaving condenser water temperature
LED	light emitting diode
LET	leaving evaporator water temperature
LVDT	linear variable differential tranducer
MUC	McQuay MicroTech Unit Controller
ORNL	Oak Ridge National Laboratory

PC	personal computer
psi	pounds per square inch
psig	pounds per square inch guage
sq ft	square feet
USACE	U.S. Army Corps of Engineers
USACERL	U.S. Army Construction Engineering Research Laboratories
USACPW	U.S. Army Center for Public Works
USEPA	U.S. Environmental Protection Agency
V	volts
VAC	volts alternating current

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